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REPORT No. 802

Balance System for the Aberdeen 13" x 15" Supersonic Wind Tunnel

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BALLISTIC RESEARCH LABORATORIES

ABERDEEN PROVING GROUND, MARYLAND

B A L L I S T I C   R E S E A R C H   L A B O R A T O R I E S

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A B E R D E E N   P R O V I N G   G R O U N D,   M A R Y L A N D

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BALANCE SYSTEM FOR THE ABERDEEN 13" x 15" SUPERSONIC WIND TUNNEL

ABSTRACT

A new design 6-component external balance system for a supersonic wind tunnel is described. The design contemplates use of Eastman electromagnetic weighing cells, and considers such factors as building vibrations and temperature changes. Interactions and their elimination are discussed, including the method of alignment of parts.

Introduction

A new flexible nozzle supersonic tunnel with 13" x 15" test section area is being built for the Aberdeen Wind Tunnels Laboratory. It was desired to provide this tunnel with an exceptionally good six-component balance. A good balance system of conventional tripod or pyramid design was first designed; but studies of the interactions produced by deflections of the struts and flexures led to a complete re-design, which is described in this report.

The balance system for this tunnel is designed for the following ranges;

Drag	$\pm$ 50 lbs (or -20 to + 80 lbs)
Lift	$\pm$100 lbs
Side force	$\pm$ 50 lbs
Pitching Moment	$\pm$600 in. lbs
Rolling Moment	$\pm$300 in. lbs
Yawing Moment	$\pm$300 in. lbs

The general design philosophy has been followed that wind tunnel balances are in general not nearly as accurate as desirable in measuring moments; plane and missile designers want center of pressure positions to greater accuracy than is usually obtainable from wind tunnel balance data. The balance was designed with the hope of obtaining an accuracy of 0.1% of actual value of each force component whenever this force component is more than 1% of maximum value which the balance can carry. This requires that the maximum value of a force component be 100,000 times the least count of the balance.

Accordingly, a number of changes were made from the originally proposed balance design in an attempt to get this high accuracy. These changes included a new design load cell, a new design balance linkage different in principle from the original pyramid design, a change in the location and design of the moment tare counterweight, use of a great deal of invar in the balance system to get rid of misalignment due to temperature differences in the various parts of the balance, the design of a vibration-absorbing foundation for the balance system, and the design of a more-efficient flexure for linkage struts.

Load Cells

The Eastman Magnetic Pot, designed by Prof. Eastman of the University of Washington (Seattle, Washington) and built by the Instrument Laboratory of Seattle was selected as the load cell, since it does have the accuracy to satisfy the requirements. This cell supports a load by the force acting on a d.c. current carrying conductor in a permanent magnet field, being similar

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in principle to the Kelvin current balance. The design is such that the permanent magnetic field across the air-gap is neither weakened nor strengthened by the variable magnetic field due to the load current, so that the load current varies linearly with the load to a high degree of accuracy.

These Eastman Pots have always been used as springs; i.e., the deflection of the load coil under load produces an electrical signal which is amplified and used to regulate the load current, so that the deflection is proportional to the applied load. The usual optical displacement sensing devices allowed as much as .010" displacement for 10 lb.load, or an equivalent spring rate of 1000 lbs per inch deflection. The deflection of the load cell causes balance misalignment and hence interactions. Accordingly, much time was spent in the improving the stiffness of these cells by a new control system.

By the use of a Schaevitz differential transformer at 5000 cycles as a displacement sensing device, the load-carrying capacity and stiffness of the Eastman pots has been increased to the point where they will now carry $\pm$ 40 lbs with a deflection of less than 0.0005", or an equivalent spring rate of above 80,000 lbs/in.

The conventional oil dash pot for damping is totally inadequate at this stiffness, so that it was necessary to add a velocity pickup, essentially a small permanent field generator which gives off an electrical voltage proportional to the velocity when the load cell vibrates. By amplifying separately displacement and velocity signals (with proper filters in each circuit), the Eastman pots have been made stable at this stiffness when connected to dummy balance linkages having many natural frequencies in the range from 20 to 500 cycles.

The development of the Eastman Pot control system with electronic damping is described in BRL Report No. 805, "A High Stiffness Electromagnetic Weighing System".

General Description of Interactions

For a 6-component balance system, the interactions can be classified as first order, second order, third order, etc. The first order, which will be called linear also, will be discussed first; these are the ones which appear when a single force (or moment) component is applied. Let F_1, F_2, ---, F_6 be the applied force or moment, R_1, R_2, ---, R_6 be the corresponding balance reading converted to lbs. (or in. lbs.); then the following set of equations holds

$$R_1 = A_{11} F_1 + A_{12} F_2 + \text{---} + A_{16} F_6$$

$$R_2 = A_{21} F_1 + A_{22} F_2 + \text{---} + A_{26} F_6$$

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$$R_6 = A_{61} F_1 + A_{62} F_2 + \text{----} + A_{66} F_6$$

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That is, when any single force component F_k is applied, all other F_j being zero, these equations give the corresponding readings R_1, R_2, ----, R_6 observed. For a correctly adjusted balance, the matrix A should be a unit matrix;

$$A_{ij} = \begin{cases} 0 \text{ if } i \neq j \\ 1 \text{ if } i = j \end{cases}$$

Since there are 30 linear interaction coefficients A_{ij} with $i \neq j$, it is plain that at least 30 separate adjustments to the geometry of the force decomposing linkage must be made to eliminate these linear interactions.

Perhaps a clearer notation is to let R_{ij} be the reading in lbs. for the ith force component when the jth component of force is applied; then in terms of the matrix A_{ij}, the following 36 equations hold

$$R_{ij} = A_{ij} F_j$$

any proper 6-component balance must permit sufficient adjustments so that the matrix A can be made a unit matrix.

The quadratic interactions may be defined by a set of equations

$$R_{ijk} = B_{ijk} F_j F_k$$

where R_{ijk} is the error in the reading of the ith component when F_j and F_k are applied simultaneously. These quadratic interactions have several causes: yielding of load cells under load, yielding of balance linkage members (stretching of struts and flexures, bending of beams and bell-cranks, yielding of ground points, stiffness of flexures), and improper location of flexures.

Since there are $6^3 = 216$ of these quadratic errors, it is rather impractical to provide this many adjustments in addition to the 30 required for the linear interactions. The quadratic interactions are made small by proper location of flexure centers, making the flexure bending stiffness very small in places where bending moment in flexure causes interaction, making flexures as stiff as possible in tension, making the base frame, beams and bell-cranks rigid, and making the load cells as stiff as practical. For example, if the two arms of a bell-crank are not exactly at right angles to the loads, when the bell-crank rotates slightly under load the ratio of arm-lengths changes, leading to a quadratic error proportional to the load times the rotational angle and hence proportional to the square of the load.

This balance system has been designed so that a great many of the quadratic interactions are zero. It was impossible to make all quadratic

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interactions zero. In particular, the design of the lift strut was made such that no moment errors appear as a result of lift applied simulatneously with any other component; as a result, there is left in the system a quadratic drag error proportional to the product of lift and drag, and another proportional to the product of lift times pitching moment. Further discussion of these interactions is better left until the description of the balance system has been given.

Angle of Attack Mechanism Fig 1

The angle of attack mechanism consists of a crescent carrying the model support strut, supported by three guide rollers in the strut carrier bracket. This allows the model angle of attack to be changed, the center of rotation being on the strut centerline and tunnel center-line, generally near the model center. The angle of attack drive is from a small motor and reduction gear to the pinion gear shaft. The strut carrier bracket also carries two pairs of rollers for the dual steel tape drive to the moment tare compensator, a fine and coarse selsyn to give remote indication of angle of attack, and a third selsyn to provide angle of attack information to the windshield drive servo-system.

The part of the crescent in the air-stream and the model support strut nearly to the model base are enclosed in a windshield which follows the strut and crescent rotation, but must never touch to produce forces; the windshield servo-drive contains a manual differ-ential adjustment to be used in case there is strut-windshield contact (indicated electrically) due to air loads bending the strut. The angle of attack of the crescent is over the range $-10^{\circ} \leq \theta \leq +15^{\circ}$, though the windshield motion is limited to a 20° range. Thus by making a windshield change, there are two available ranges, $-10^{\circ} \leq \theta \leq +10^{\circ}$ and $-5^{\circ} \leq \theta \leq +15^{\circ}$.

Moment Tare Compensator

It was originally planned to extend the crescent into the region above the air-stream, and adjust the weight W until the c.g. of the upper half of the crescent at B in figure 2a was 180° from the c.g. of the model and lower half of the crescent of A, and the products of the two weights times their corresponding lever-arms was equal, thus bringing the c.g. of the entire mass to the center of rotation; then there would be no moment tare as the angle of attack was changed.

There were two objections to this method of moment tare compen-sation. The most serious objection is that crescent deflection causes first order interactions of considerable magnitude. The crescent deflects under air loads applied to the sting; the worst deflections are caused by a side force, which applies combined torsion and bending in the weaker direction to the crescent. Under this deflection, the heavy counterweight at the end of the upper crescent horn is deflected out of the vertical plane through the tunnel axis and hence produces a

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rolling moment. This leads to a first order interaction, rolling moment produced by side force. For one position of the crescent, this interaction could be eliminated by intentional misalignment of the points where side force and rolling moment forces are taken of the floating beam. But since the crescent gets stiffer as angle of attack is increased, this correction would hold at only one angle of attack. Secondly, since all models will not have their c.g. on the support sting centerline, the weight at the end of the upper crescent horn would have to be adjustable in two directions, parallel and perpendicular to the sting centerline. Space available in the design and the difficulty of getting power leads to a motor drive for the counterweight adjustments were also arguments in favor of a new moment tare compensator design.

Accordingly it was decided to change the method of moment tare compensation. Essentially the change consists of removing the upper half of the crescent and weight W of figure 2a, and hanging it below the crescent so that it revolves about an effective center C' as in figure 2b; a dual steel tape drive makes the angular displacement of the moment tare compensator about its effective center C' equal to that of the crescent, strut and model about C. Then if the weights W_1 and W_2 are adjusted until the vector $W_o \times AC$ is equal to the vector $W_c \times C'B$ in magnitude and direction (where W_o, W_c are the weight of the crescent assembly and the compensator assembly respectively), perfect moment tare compensation will be achieved with no air forces.

It will be noted that the tape tension increases as the angle of attack becomes negative, which would stretch the tape more and tend to make the compensator rotate faster than the crescent. This error is compensated to the first order by making the tape radius on the compensator vary slightly greater than on the crescent.

In studying the adjustments of weights W_1 and W_2, it is easier to visualize the problem if the moment tare compensator is considered to translated back to the configuration of figure 2a. Then it can be seen that the weight W_2 which moves perpendicular to the strut does not need to be readjusted whenever the weight of the strut and model are changed, as long as the strut and model center of gravities are on the strut centerline. Hence the adjustment of W_2 is manual, and will require opening the balance tank. When W_2 is properly adjusted, the c.g. of the whole assembly in figure 2a is on the strut centerline. When the strut and model combination is changed, weight is only added or subtracted on the strut centerline, which will move the c.g. of the assembly along the strut centerline; the c.g. can be brought back to the center of rotation by changing the position of W_1 which moves parallel to the strut centerline.

Since W_1 will have to be readjusted whenever the model is changed, it rides on a motor-driven screw and hence is adjustable by remote control.

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Moment Table

The moment table is shown in figure 3; it, with the crescent carrier bracket assembly bolted to its top carrying the crescent, strut and model, constitutes the floating part of the balance system. The moment table carries tracks on which the moment compensator rides, so that this compensator has an effective center of rotation at C', the center of curvature of the tracks.

The moment table is restrained by 7 struts ending in flexure connections, whose lines of application are shown by the 7 numbered arrows. Struts 1 and 2 take care of drag and pitching moment, 3 and 4 handle side force and rolling moment, 5 is for lift, while a couple applied through 6 and 7 resists yawing motion.

Force Decomposition

The struts which support the moment table and hence the whole floating part of the balance system (including crescent carrier bracket, crescent, model strut and moment tare compensator) are all parallel or perpendicular to the tunnel axis, and all horizontal except for the lift strut which is vertical. These struts have flexures at each end which are specially designed to have very low bending moments, yet be stiff in compression or tension. Neglecting the residual bending moments, and assuming the struts are properly aligned, the 6 components of forces are already partially resolved in this fashion.

Forces	Struts resisting the forces
Drag, Pitching Moment	No. 1, 2
Side force, yawing moment	No. 3, 4
Lift	No. 5
Yawing moment	No. 6, 7

Since the struts 6, 7 are parallel to struts 1, 2, this decomposition is not evident until one is informed that the linkage at the far end of struts 6, 7 is such that these can only transmit a torque; i.e., when 6 is under tension, 7 is under a numerically equal compression.

Yaw Linkage Mechanism

The principle of the yaw linkage is shown in figure 4. The strut No. 6 ends in a flexure on the reversing beam, which has crossed flexures to ground at its center and is connected at its other end with crossed flexures to a floating beam. Strut No. 7 from the moment table is connected to the other end of the floating beam, and the yaw take out strut is attached by flexure to the center of the floating beam.

Thus a compression load of F lbs. applied to the yaw takeout strut results in a tension of F/2 lbs in strut 7 and a compression of F/2 lbs in strut 6, and hence applies a pure torque of magnitude Fb/2 to the moment table. Of if M_y yaw moment is applied to the moment table a force

$$F = 2M_y/b$$

appears at the yaw takeout strut.

Neglecting flexure stiffness, the moment table can be displaced in a direction parallel to struts 6, 7 without causing force or motion to occur at the yawing moment takeout strut; both the floating and reversing beams rotate about their centers under this displacement.

The yawing moment takeout strut leads to a bell-crank which changes the force direction to vertical at a convenient location for the force measuring cell. The bell-crank arm lengths ratio is selected so that the full load on the measuring cell (± 35 lbs) corresponds to the desired maximum moment specified in the design.

It now becomes evident that the other struts 1, 2 parallel to the tunnel axis will carry loads produced by drag and pitching moment.

Drag - Pitching Moment Resolving System

The principal of this linkage is shown in figure 5. Strut No. 1 is a distance a below the tunnel centerline (center of moments) and extends to a flexure at the upper end of the reversing beam. Strut No. 2 is a distance $2a$ below the center of moments, and extends to the lower end of the floating beam. The reversing beam is grounded by a crossed flexure hinge at its center, and a crossed flexure hinge connects its lower end to the upper end of the floating beam.

The drag takeout strut is connected at a third of the way down the floating beam, and the pitching moment takeout strut is connected to the center of the floating beam, both by flexures. Using the plus sign for compression and the minus sign for tension in struts, a drag force of D lbs. and a moment of M in. lbs. produces in the struts the following forces:

Strut No. 1	$2D + M/a$
Strut No. 2	$-D - M/a$
Drag takeout strut	$-3D$
Moment takeout strut	$-2M/a$

Investigation shows that the heights of the drag and moment takeout struts must be correctly located to within about 0.0001" in order to get rid of interaction between these two forces. Hence special fine adjustments

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are included in these heights.

Referring to figure 5, the drag and pitching moment takeout struts are about 3.5 inches apart, and a force of $3D = 240$ lbs. is applied to the drag takeout strut when full load drag is applied. If this height is off by an amount E inches, there is a resulting force applied to the moment takeout strut of $240E/3.5$. The least count in high range pitching moment is 600 lbs. $\times 10^{-4} = .06$ in. lb., and since the distance A is about 28", the resulting force $2M/a$ in the moment takeout strut is $0.12/28$ lbs. Equating these values and solving for E gives

$$E = (.012)\,(3.5)/(240)\,28)$$

$$= 0.62 \times 10^{-4} \text{ inches}$$

That is, to eliminate linear moment error due to drag, the height of the drag takeout length must be adjusted to within 62 microinches.

The height of the moment takeout strut requires a similarly fine adjustment. The schematic principle of these fine adjustments is shown in the small sketch in Figure 5.

The drag and pitching moment takeout struts lead to bell-cranks which serve to change the forces to vertical forces of ± 35 lbs range which acts on the drag and lift cells. The nominal drag range is ± 50 lbs. The drag bell-crank carries provisions for adding a pre-load which is equivalent to -30 lb drag; this gives a range of -20 to +80 lbs. Since these pre-load weights are in the balance tank, subject to varying pressure and hence varying buoyancy, a hollow sphere of the proper volume is used to counter-balance the buoyancy effect. Without the sphere, the error due to buoyancy change when the balance tank is evacuated amounts to about ten least counts of the drag balance.

Since the c.g. of the floating system is evidently down-stream of the center of moments, the pitching moment bell-crank carries an adjustable counter-weight to balance out the resulting tare moment, so that the moment cell will receive only the aerodynamically caused moment.

Side Force - Rolling Moment

Struts 3 and 4 from the moment table lead to a resolving linkage which is identical in principle to that of the drag force-pitching moment. The side struts 3 and 4 are slightly closer to the tunnel centerline, so that the flexures at the moment table and of struts 1, 2, 3, 4 can all be located on a vertical line through the center of moments. This location of the flexures eliminates one source of force interactions.

The floating assembly is practically symmetric about a vertical plane through the tunnel centerline; hence only small counterweights have to be used, mostly to balance the bell-cranks themselves, so that the force-measuring cells will receive no tare loads.

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Lift Takeout Linkage

Lift could have been taken out simply by connecting a vertical strut from the moment table to the lift cell, this strut being directly under the moment center. However, when drag and moment are applied, the moment table is going to be displaced and rotated in the lift-drag plane; the lift strut would no longer be vertical, and there would be both drag and moment errors proportional to the products LD and LM. Drag accuracy is not regarded as highly important in supersonic wind tunnel testing, since the proportion of drag due to skin friction, depending on Reynold's number and surface roughness, cannot generally be scaled properly; but high accuracy in pitching moment is frequently desired. According, a design of lift strut was selected which eliminates (to a first order of approximation) the quadratic interactions on pitching moment and rolling moment proportional to the product of lift times either drag, side force, pitching moment, or rolling moment.

There would be displacement of the floating systems under loads due to elastic deflections of all parts, even if the load cells did not yield at all with applied loads.

The lift strut system is shown schematically in figure 6. The moment table is distorted in order to show the floating links AD and BDE which are inside it. The moment table is connected by a flexure at B to the top of the upper floating link, free to bend in either direction. The moment table is connected by flexure strips to the bottom of the lower floating link, at E, in such a way that E can only be displaced vertically with respect to the moment table.

To understand how this system operates, it is simplest to consider what happens if the moment table is held ridgidly fixed, and the upper end B of the lower floating link is disconnected from the lift beam and moved to B'. The line diagram to the right in Figure shows the motion of the flexure-points relative to the moment table. When B is moved to B', point E moves upward slightly to E; point D moves on a circular arc about A to the point D', rising a distance $\ell_1(1 - \cos \delta)$. The vertical component of length DB changes by an amount $\ell_3(1 - \cos \beta)$. The resulting motion of B to B' is the same as if B' moved on a circular arc of radius r about a point C', up to terms in the 4th power of the angle x. The relative lengths ℓ_1 ℓ_2 and ℓ_3 were so chosen that the effective center C is at the center of moment of the balance system. The motion of B relative to the moment table is not restricted to one plane; hence B moves on a spherical surface of radius r about the center of moments C, when motion of B relative to the moment table is considered.

This geometrical relationship, that the distance BC = r remains constant, means that as far as resolution of forces is concerned, the whole lift linkage can be considered equivalent to a simple strut attached to the lift beam at B and attached to the balance system at the center of moments.

Hence it results that when the moment table is displaced by combined loads,

the reaction of the lift strut on the moment table produces no pitching, yawing or rolling moments. Lift does produce drag and side force errors under combined loads which displace the moment table, but these are small because the equivalent lift strut from B to C is quite long.

The flexure B connects to a lift beam, the other end of which carries counterweights to balance out the dead weight of the floating system; this long end of the beam also connects by strut to the lift cell.

Flexures

For flexures which have to bend in both directions and also twist slightly, a special design was evolved at the Wind Tunnel. This design bends very much more easily, yet is stiffer in tension and compression, than a flexure formed by a necked down portion of rod. With regard to these properties it is equivalent to two flat flexures, arranged at 90° but placed in series, a design which is often used. But the new design puts the flat flexures in the same axial position on the strut. All these features aid in reducing force interactions in the balance. This flexure is illustrated in figure 7. Figure 7b illustrates the consequence of not having a strut flexure on an axis through the center of moments. This is a plan view of a strut to the floating system to resist drag. If the floating member rotates to the dotted position due to applied yaw, the compression in this strut caused by drag will produce quadratic errors in yaw moment and side force proportional to the product of drag times yaw moment.

Most of remaining flexures which have to bend in one direction only are crossed flexure hinges.

Installation of Balance

The balance system is made so that it can be quickly removed or inserted in the balance tank. The crescent carrier bracket which normally is bolted and doweled to the top of the moment table, can also be fastened in its working position to the ceiling of the balance tank, so that models with internal strain gage balances can be tested when the regular balance system is not installed. To install the regular balance system, the crescent carrier bracket is raised slightly, the balance system is rolled (one side of the balance tank having been removed) and roller jacking screws loosened. The balance thus is lowered so that it rests on two ways similar to those of a lathe carriage. Three capscrews fasten it down to these ways. The crescent carrier bracket is lowered onto the moment table and bolted, and electrical connections to the load cells are plugged in.

Displacement Sensing on the Moment Table.

If the displacement sensing devices in the load cells are used, the load cell coil itself will yield less than 0.001" under full load. But the deflection of the model under air loads comes from strut deflection,

crescent deflection, and those of the strut carrier, moment table, struts to the moment table, reversing beams, ground points, floating beams, struts from floating beams to bell-cranks, bell-crank deflections, struts from bell-cranks to load cells, in addition to the deflection of the load cell itself.

To reduce the total deflection of the model under load, the moment table itself is provided with 6 displacement sensing Schaevitz transformers. Six signals from these are taken to a mixer, from which displacement control signals are obtained for the 6 load cells. Generally two moment table displacement signals have to be mixed in the proper ratios to get the signal for controlling the current to each load cell.

This method of control should hold the moment table itself practically fixed, except for the displacement of not over 0.001" required to get an adequate displacement signal from the six Schaevitz transformers. Since the introduction of many elastic members between the load cells and the (generally two) displacement pick-ups which controls them greatly changes the nature of a portion of the servo-loops involved, it is not certain that this method control will work. It has been made to operate fairly well on a 2-component mock-up, and the geometry of the balance pretty much makes the six force components independent in pairs. Hence it seems probable that the system will work.

Adjustments required during balance alignment to remove interactions will probably be slightly altered if control is swtiched from controlling the load cell displacements to controlling by moment table displacements.

Since the model support strut will be frequently changed and hence will vary in flexibility, it is recommended that calibration be done with a very rigid replacement for the model support strut. At the completion of calibration, interactions which cannot completely be removed should be measured and recorded. Then additional increments of interactions due to bending of the model support sting can be calculated and corrected in the course of computations, assuming that model strut deflections are measured for each model strut used.

Angle of Attack Control

The crescent carries a curved rack and is driven by a pinion gear. The rack is 43.2" P.D. and the pinion is 1.2", giving a gear ratio of 36:1. The pinion shaft is driven by a Bodine Geared Reducer Motor 1725 R.P.M. with a gear ratio of 48:1. The motor is 60 cycle, 220V, 3ϕ, 1/70 h.p. This will drive the crescent at a speed of $\frac{1725}{36 \times 480}$ r.p.m. or $\frac{1725 \times 360}{36 \times 480} = \frac{1725}{48}$ degrees/min. approximately 36 degrees/min. The resulting time per degree is 60/36 = 1.67 seconds per degree; hence the whole range of 20° can be covered in about 34 seconds. The drive gear teeth are 10 pitch, 20° stub teeth.

The pinion shaft drives a fine and coarse selsyn pair for remote indication of angle of attack, and a selsyn for the windshield follow-up servo. The signal from the windshield servo goes through a differential selsyn, which enables the windshield to be adjusted manually in case of strut-windshield interference.

Temperature Effects

If the temperature of the whole balance were to change, with all parts undergoing the same temperature rise, and if all parts were made of material with the same coefficient of expansion, the performance of the balance would not be affected by temperature changes. However, when the tunnel is started, the walls of the air passage usually drop about 20°F. below room temperature, in spite of the fact that T_o is kept about 20° above room temperature. The parts of the balance system up in the windshield and the parts just under the tunnel floor will get cool first, while the parts near the bottom of the balance tank will cool more slowly, and never cool off as much as the tunnel walls cool.

Thus, temperature gradients will exist in the parts of the balance system, and this will result in balance system misalignment if the parts have temperature coefficients of expansion different from zero. For example, if the upper half of the reversing beam (assumed to be steel) in figure 5 became 1°F. cooler than the lower half, 30 lb. applied drag would result in .03 in.lbs. of moment. Since the least count of the moment balance is 0.01 in. lbs, this effect would be serious. For this reason, all those parts of the balance system which would cause appreciable errors due to temperature changes are being made of invar.

Effect of Vibrations

Foundation vibrations will also have the effect of producing vibrations in the parts of the balance system. For example, the floating part of the system (moment table, crescent, strut, moment tare compensator) are counterbalanced by a heavy counterweight at the other end of the lift beam. Vertical vibrations of the foundation would cause bending vibrations in this lift beam; since the electromagnetic pot to balance and measure applied forces is governed by the displacement pickup, and exerts forces proportional to relative displacement pickup, the electromagnetic pot would exert forces trying to prevent relative motion at the displacement pickup. These forces would then appear on the electrical weighing signal, and have to be filtered to obtain a readable signal. The larger the vibrations, the more elaborate is the filter needed; this filter has the objection of slowing down the response of the system to applied forces, as well as increasing the expense and complexity of the system.

In a similar manner, horizontal vibrations would cause vibrations in the parts of the balance system, and oscillations in the electrical force signals. Also, rotational vibrations would have similar effects. Hence measurements were made of vertical and horizontal (in two directions) vibrations and of rotational vibrations about three axes, using seismo-

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graphs recording on a Brush recorder. Vibrations of about 0.0005"
amplitude occur on the foundations of the present bomb tunnel
foundations. The main sources of the vibrations were found to be the
main motors (20 cycles), the intake compressors and large vacuum pump
(4 to 5 cycles per second) and the air conditioning compressors (about
10 cycles). The large centrifugal carrier compressors would cause 90
cycle vibrations, but these were not detected, hence must be small.

For horizontal vibrations, the inertia of the floating system adds
apparent oscillatory drag and side forces. For vibrations of amplitude
A and frequency w, the acceleration amplitude is Aw^2, and the resulting
force is MAw^2. The floating system will weigh about 200 lb.; the result-
ing force from an amplitude of .0005" is

$$F = MAw^2$$

$$= \frac{200}{32} \times 5 \times 10^{-4} \times (2\pi \times 20)^2$$

$$= \frac{10}{32} \times (4\pi)^2$$

$$= 50 \text{ lb. approximately}$$

This is the magnitude of the oscillating force which would appear on the
weighing system if it were infinitely stiff. While the weighing system is
not infinitely stiff, 50 lb. drag would deflect it only about 0.0001",
hence this amplitude of vibration of the base frame would actually indicate
an oscillating drag of about 40 lbs. Vertical masses are counterbalanced
so that vertical vibrations do not produce this kind of effect.

Two steps were taken to cut down anticipated difficulties. The firm
of Sandberg-Serrell, Inc., were asked to make the parts of the balance
system sufficiently stiff to keep all resonance frequencies above 20 cycles,
and to provide isolation of the balance from foundation vibrations as far as
possible.

For vibration insulation, a deep pit is to be dug under the balance tank.
The balance frame will have three supports, through holes sealed by flexible
bellows in the bottom of the balance tank, to a massive concrete block. This
block in turn is supported on four posts or columns fixed at their lower ends
in the foundation. These posts are sufficiently rigid so that air forces on
the model will not cause serious model displacement; and the block is heavy
enough to bring the natural frequency of the balance system foundation down
to a low value, below the 20 cycle frequency.

Acknowledgements

The force component resolving linkage was designed in detail by
Sandberg-Serrell, Inc. of Pasadena, following preliminary sketches and
a 1/4 scale mock-up produced by the Aberdeen Wind Tunnel. Credit for
the reversing and floating beams type of linkage belongs to Robert Smith,
who rediscovered this principle and analyzed it while he was a Wind
Tunnel employee.

The development of the control and damping system for the Eastman
Magnetic pots used as load cells was done by Gene Slottow. The load
cells and their electronic control equipment was built by The Instrument
Laboratory of Seattle, Washington. The Bethlehem Shipbuilding Company,
fabricators of the tunnel, are making the working drawings and fabricating
the balance system. The angle of attack drive and windshield follow-up
servo have been built by Hanson-Gorrill-Brian, Inc., of Glen Cove,
New York.

Concluding Remarks

The desirable features for a supersonic wind tunnel balance are high
accuracy extending down to small force measurements, ease of adjustments
to eliminate interactions, accessibility of parts for adjustment or replace-
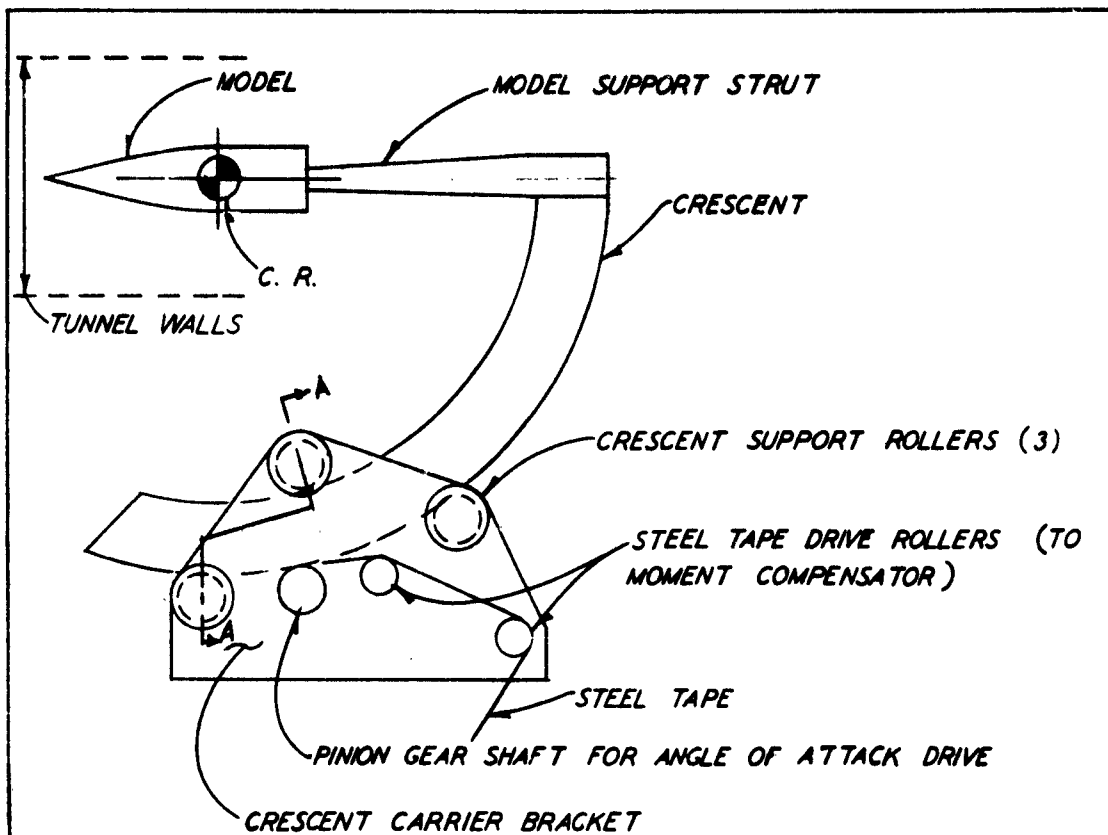ment, compactness, and reliability.

Compactness is wanted because the balance tank must be pumped down to
model base pressure, which may be as low as 5 millimeters absolute. Because
of the wide changes in tank pressure, grounded points of the balance system
must not be moved by tank wall deflections due to pressure changes. Because
of the wide temperature variations which occur when the tunnel is started,
it was found necessary to use a good deal of invar for balance parts. A
study of foundation oscillations showed the need of balance vibration
isolation in horizontal directions.

A study of methods of adjustment to eliminate interactions (discussed
in the appendix) indicated that the unorthodox rectangular arrangement of
struts and reversing levers is easier to adjust than the more usual tripod
system first considered; also this design permitted the large electro-
magnetic pots or load cells to be placed inside the balance tank; in the
original tripod system, these were to be placed outside, with special design
bell cranks intended to bring out forces thru sealed joints which it was
hoped could be made friction-less and not influenced by pressure. The
electromagnetic load cells were tested in a pressure of 5 millimeters
absolute to check that they would not run hot under continued loading.
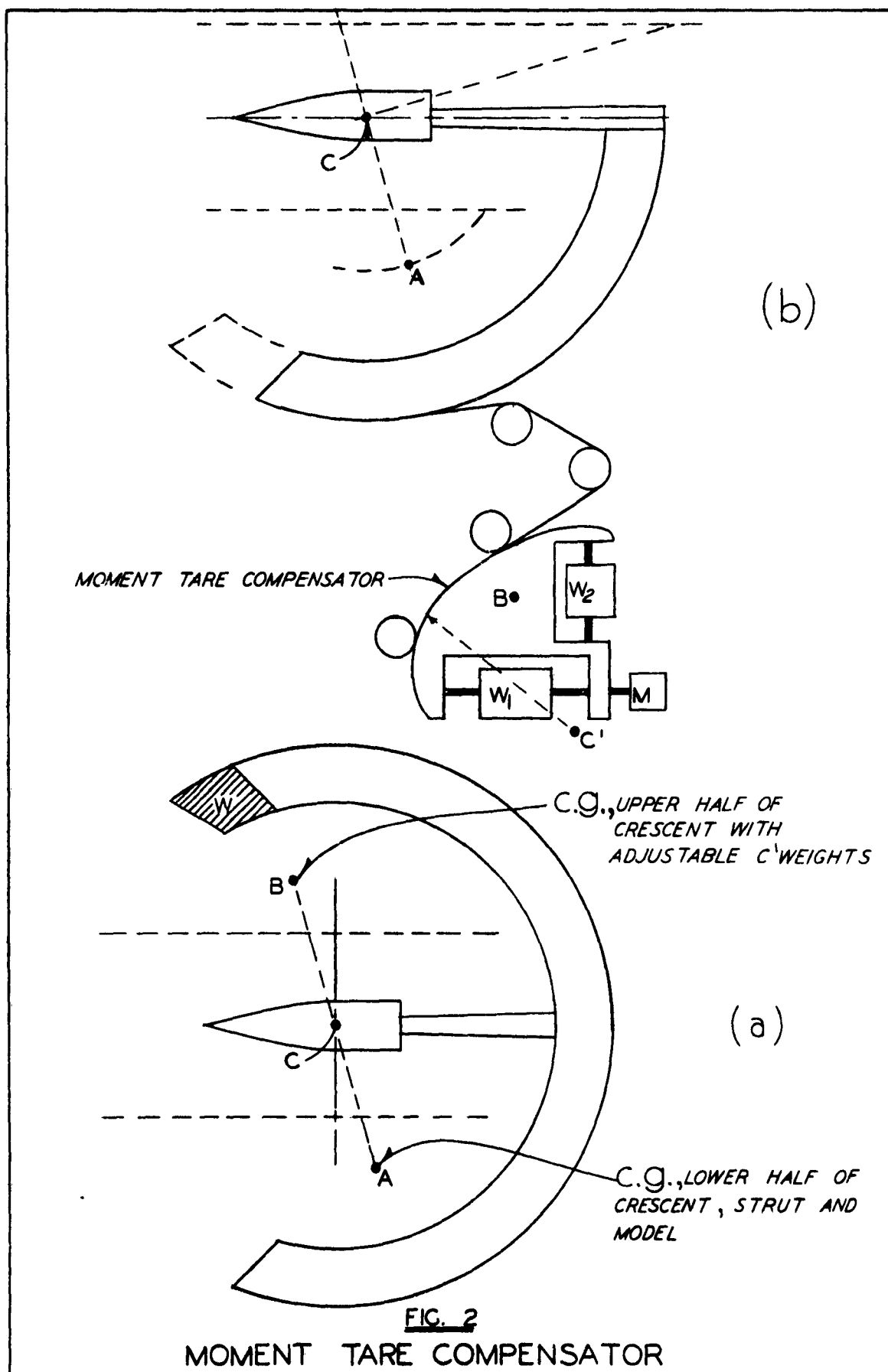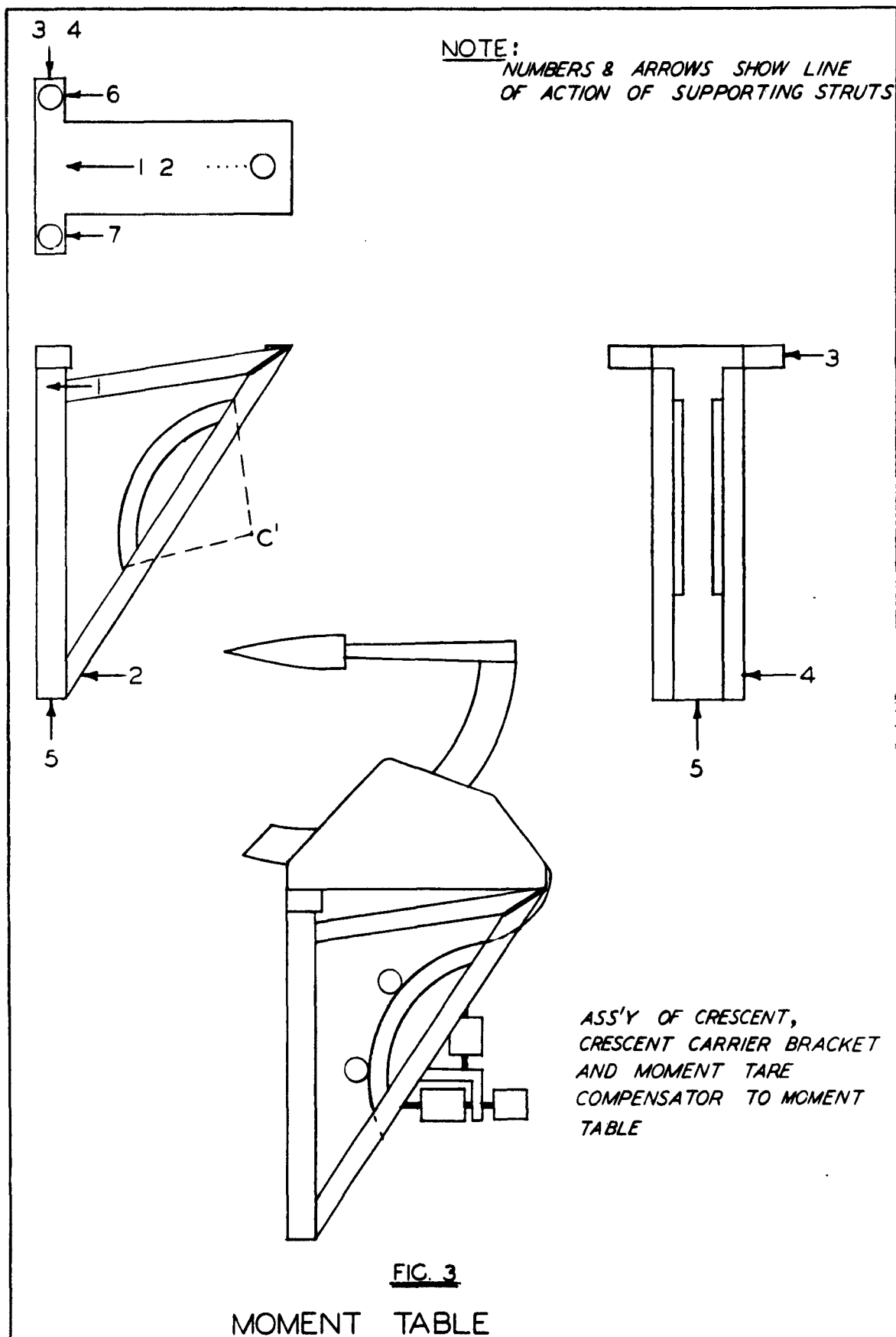
T. L. Smith

TURNER L. SMITH

MODEL

MODEL SUPPORT STRUT

CRESCENT

C. R.

TUNNEL WALLS

A

CRESCENT SUPPORT ROLLERS (3)

STEEL TAPE DRIVE ROLLERS (TO MOMENT COMPENSATOR)

A

STEEL TAPE

PINION GEAR SHAFT FOR ANGLE OF ATTACK DRIVE

CRESCENT CARRIER BRACKET

ROLLER

CRESCENT

GEAR SECTOR FOR DRIVE

STEEL TAPES WRAP ON THESE FLATS

ROLLER
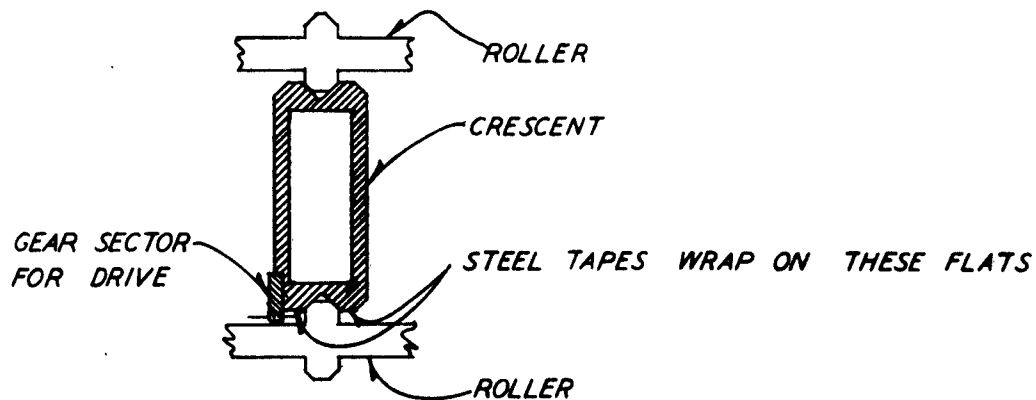
SECTION A-A

FIG. 1

ANGLE OF ATTACK MECHANISM

(b)

MOMENT TARE COMPENSATOR

W_2

B•

W_1

M

•C'

C.g., UPPER HALF OF
CRESCENT WITH
ADJUSTABLE C'WEIGHTS

W

B•

C

(a)

A

C.g., LOWER HALF OF
CRESCENT, STRUT AND
MODEL

FIG. 2

MOMENT TARE COMPENSATOR

NOTE:
NUMBERS & ARROWS SHOW LINE
OF ACTION OF SUPPORTING STRUTS

ASS'Y OF CRESCENT,
CRESCENT CARRIER BRACKET
AND MOMENT TARE
COMPENSATOR TO MOMENT
TABLE

FIG. 3

MOMENT TABLE

NO. 6 STRUT

REVERSING BEAM

b

NO. 7 STRUT

YAW TAKE-OUT STRUT

FLOATING BEAM

FIG. 4

LINKAGE FOR MEASURING YAW

D
P.M.

a

FORCE = - 3D
DRAG TAKE-OUT STRUT

+ 2D + $\frac{M}{a}$
STRUT NO.1

SPHERE

REVERSING BEAM

A

FLOATING BEAM

P.M. TAKE-OUT STRUT
FORCE = - 2 $\frac{M}{a}$

-D - $\frac{M}{a}$
STRUT NO. 2

DRAG CELL

P.M. CELL

FINE ADJUSTMENT

DRAG TAKE-OUT STRUT

COARSE ADJUSTMENT

PRINCIPLE OF DRAG TAKE-OUT
HEIGHT ADJUSTMENT
DETAIL "A"

FIG. 5

DRAG - PITCHING MOMENT RESOLVING LINKAGE

24
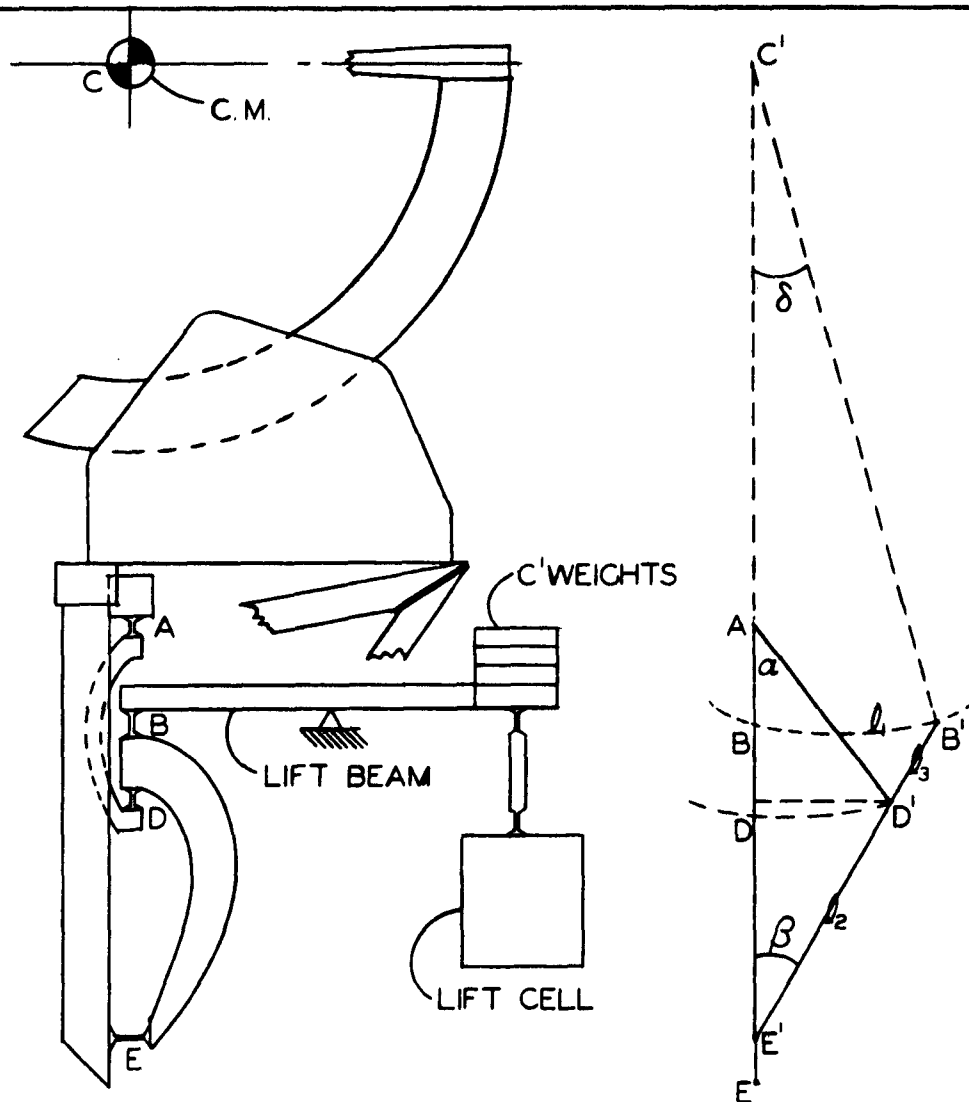
SMALL ANGLE APPROXIMATIONS

$$\ell_1 \alpha = \ell_2 \beta$$

INCREASE IN HEIGHTH OF D' FROM $D = \ell_1 \frac{\alpha^2}{2}$

DECREASE IN VERTICAL PROJECTION OF $D'B' = \ell_3 \frac{\beta^2}{2} = \ell_3 \frac{\ell_1^2}{\ell_2^2} \frac{\alpha^2}{2}$

INCREASE IN HEIGHTH OF B' BY DIFFERENCE $= \left[\ell_1 - \ell_3 \left(\frac{\ell_1}{\ell_2}\right)^2\right] \frac{\alpha^2}{2}$

IF B' REMAINS AT CONSTANT DISTANCE Γ FROM C,
INCREASE IN HEIGHTH OF $B' = \frac{\Gamma \alpha^2}{2}$

ALSO $\left(\ell_2 + \ell_3\right)\beta = \Gamma \delta$

EQUATING HEIGHTH INCREASES OF $B' = \frac{\Gamma \delta^2}{2} = \frac{\alpha^2}{2}\left(\ell_1 - \ell_3 \frac{\ell_1^2}{\ell_2^2}\right)$

$$\Gamma \frac{\left(\ell_2 + \ell_3\right)^2 \beta^2}{\Gamma^2} = \alpha^2 \left(\ell_1 - \ell_3 \frac{\ell_1^2}{\ell_2^2}\right)$$

FIG. 6
LIFT STRUT SYSTEM

(a)

section A-A

C.M.

(b)
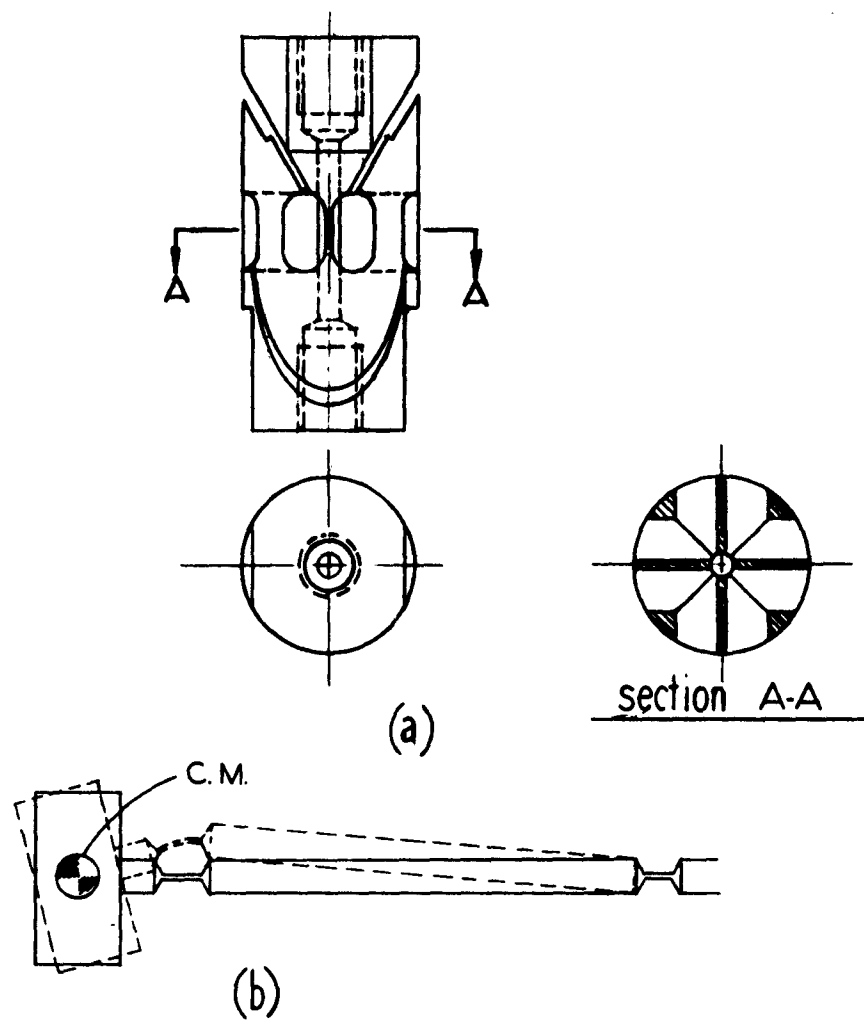
FIG. 7

SPECIAL FLEXURE DESIGN

Appendix: 13" x 15" Tunnel Balance Calibration Procedure.

Axis Conventions. For the purpose of this calibration description, the following conventions are adopted.

> x axis points upstream
>
> y axis points vertically upward
>
> z axis to the right when observer is looking upstream

This forms a right-hand system of coordinate axes (see Fig. 1). Direction of positive moments is the right-hand rotation sense about the corresponding axes. Thus positive pitching moment is a couple causing the nose of the model to rise, positive roll is right wing down, and positive yaw is a turn to left.

> Positive rotation of 90° about x axis carries y into z axis
>
> Positive rotation of 90° about y axis carries z into x axis
>
> Positive rotation of 90° about z axis carries x into y axis

(This reverses the directions of x and y axes and of yawing moment from the convention that has been used at Aberdeen. But the conventions used at Aberdeen are a right-hand axis system, in which the conventions for roll and pitch are left-hand while yaw is right-hand. This mixed system is confusing to use in analysis.)

Design Range (High range)

Forces in pounds

> F_x: ±50 or 20 to -80 (negative of drag)
>
> F_y: ±100 (lift)
> F_z: ±50 (side force)

Moments in inch pounds

> M_z: ±600 pitching moment
>
> M_x: ±300 rolling moment
>
> M_y: ±300 yawing moment

Each component has a low range of 10% of the corresponding high range; range changing is by means of a range switch on the corresponding Brown Instrument. The least count of the Brown Instruments is 0.01% of the range.

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Adjustment for Mechanical Alignment

For this purpose the base frame will need to have:

     1. Surface for precision leveling, ground flat and provided with a protective cover plate.

     2. Three reference surfaces, perpendicular to the three co-ordinate axes x, y, z, of known x, y, z coordinates respectively to measure from to establish a coordinate system.

     3. Two reference lines, parallel to the x and z axes, to enable transits to be set up so that their telescopes swing in the xy and zy planes. (The reference line parallel to the z-axis to be parallel to the v-groove for the support rail.)

Procedure for assembly.

     1. Assemble the moment table, crescent support bracket, crescent, strut, and a strut extension which is a light, long rigid tube of uniform diameter and straight.

     2. Support this assembly (See Fig. 2) on the previously leveled base frame by the adjustable deflection limit stops. By adjusting these stops, fasten the floating assembly in the correct position, using transits.

     3. Assemble the lift linkage, adjusting length ℓ_{19} so that the lift takeout lever is level.

     4. Assemble the drag-pitching moment system, adjusting ℓ_{1-2} so that the reversing beam is vertical and adjusting ℓ_{3-4} so that the floating beam is vertical. Adjust ℓ_5 and ℓ_6 so that the bell-crank arms are vertical and horizontal.

     5. Make similar adjustments in assembling the side force-rolling moment system.

     6. Similar adjustments on the yawing system.

     7. Adjust tape length on moment tare compensator so that counter-weight shaft is horizontal at zero angle of attack.

     8. Mount external differential transformers and adjust their push rod lengths until their cores are centered; make necessary phase balance. Requires 5000 K.C. oscillator and oscilloscope (C.R.).

     9. Put in pots, adjust for center position of coil by means of the signal from the differential transformer in pots.

     10. Make rough adjustments to potentiometers so that Brown Instruments read forces approximately correctly.

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Elimination of Interactions

1. Lift

Description of linkage (See Fig. 5)

The linkage consists of the moment table M, a load transmitting link A, a second link B, and the take-out lever L with counterweight. Horizontal flexures at ⑰ permit the lower end of link B to have a vertical motion.

The ratios of lengths in links A and B are so selected that if the moment table M is held fixed and ⑱ is disconnected from the lever L (See Fig. 7), the end ⑱ of B moves (approximately) on a spherical surface about a center C at the height of the tunnel axis. It then follows that this linkage is equivalent to a simple strut from ⑱ to C. This virtual center C is on the extension of the line through ⑰ and ⑱ , and adjustment consists in making the line through ⑰ and ⑱ pass through the moment center, and in making the line from ⑱ through C vertical.

Adjustments

Apply a lift load at the center of moments; read pitching moment, rolling moment, drag and side force.

a. Make x correction at either ⑰ or ⑲ to get zero pitching moment due to lift. Equations are approximately

$$x_o = x_{19} + (x_{19} - x_{17}) = 2x_{19} - x_{17}$$

$$PM = L\,x_o = L\,(2x_{19} - x_{17})$$

since approximate $y_{17} = 2y_{19}$. Hence adjustments at ⑲ must be twice as fine as at ⑰ .

b. Similarly make a correction at either ⑰ or ⑲ to correct rolling moment. The rolling moment error due to lift L is

$$RM = (-2z_{19} + z_{17})\,L$$

c. The equivalent strut from ⑱ to C is about 40" long. From the two preceeding adjustments to obtain zero roll moment and zero pitch from lift, we can now assume that $x_o = z_z = 0$. Then the drag error due to lift is

$$D = -Lx_{18}/40"$$

Adjust x_{18} to make this zero.

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d. Similarly adjust z_{18} to reduce to zero the side force due to lift, which is

$$S.F. = + Lz_{18}/40$$

e. With these adjustments properly made, there is no way in which lift can produce yawing moment.

Drag - Pitching Moment Adjustments

Referring to Figure 3, a drag D puts 2D compression in strut 1 - 2 and D tension in strut 3 - 4. Pitching Moment P puts P/28.5 compression in strut 1 - 2 and P/28.5 tension in strut 3 - 4. The resulting errors due to misalignment are:

(1) $SF = (2D + P/28.5) (z_1 - z_2) + (D + P/28.5) (z_4 - z_3)$

(2) $YM = (2D + P/28.5) z_1 - (D + P/28.5) z_3$

(3) $L = (2D + P/28.5) (y_1 - y_2) + (D + P/28.5) (y_4 - y_3)$

(4) $RM = 28.5 (2D + P/28.5) \dfrac{z_2 - z_1}{x_1 - x_2} + 57 (D + P/28.5) \dfrac{z_3 - z_4}{x_3 - x_4}$

Method of Procedure. Apply a drag, then a pure pitching moment (or apply a drag, and a combined drag and pitching moment), and read SF, YM, L, RM. Since the lift system has been aligned, the pitching moment could even be obtained by hanging a weight on the strut, distant from the moment center.

First, use equation (2), which given two equations,

(2a) $YM_D = D (2 z_1 - z_3)$

(2b) $YM_P = (P/28.5)(z_1 - z_3)$

These two equations determine the errors in alignment z_1 and z_3.

Next use equation (1), which for the two loadings gives

(1a) $SF_P = 2D (z_1 - z_2) + D (z_4 - z_3)$

(1b) $SF_P = (P/28.5)(z_1 - z_2) + (P/28.5)(z_4 - z_3)$

Since z_1 and z_3 have been found, these two equations determine z_2 and z_4.

From equation (3) the two loadings give

$$(3a) \quad L_p = 2D\,(y_1 - y_2) + D\,(y_4 - y_3)$$

$$(3b) \quad L_p = (P/28.5)(y_1 - y_2) + (P/28.5)(y_4 - y_3)$$

These equations determine the errors in $(y_1 - y_2)$ and $(y_4 - y_3)$. Correct these by changing y_1 and y_3; for changing y_4 and y_2 would mess up the drag-pitching moment separation.

It will be noted that if these corrections are made, then there can be no rolling moment error, since equation (4) will be satisfied, with $z_1 = z_2 = z_3 = z_4 = 0$*.

Finally y_5 is adjusted so that no moment appears when drag is applied (using first coarse, then fine adjustment) and y_6 is adjusted so that no drag appears when pitching moment is applied.

If preferred, equation (4) can be used instead of equation (2); then the interactions of drag and pitch on roll moment can be made exactly zero, but a small yawing moment may appear due to drag and pitching moment.

Side Force - Rolling Moment Adjustments

This system is identical in adjustment method with the system just discussed in the last paragraph. The equations are (See Fig. 4):

---

* Actually, due to flexure stiffnesses, z_1, z_2, z_3, z_4 will not be quite zero. Remember that when drag is applied, the moment table moves downstream by perhaps .001" for 50 lb. drag. Similarly, under a pitching moment the moment table rotates slightly. This produces bending in the struts to the side force-rolling moment system, and the bending of these produces yawing motion. Actually, therefore, z_1 and z_2 will be slightly different from zero to compensate for this flexural stiffness. This need not concern the person making the adjustments; as it is automatically accomplished; but it may mean that drag and pitching moment produces a slight rolling moment reading. Nothing can be done about this.

This effect due to yielding of the moment table (under loads) and flexure stiffness, will be different when displacement control is located in the pots instead of between the moment table and base frame.

(1) $\quad D = (2SF + RM/28.75)(x_8 - x_7) + (SF + RM/28.75)(x_9 - x_{10})$

(2) $\quad TM = (2SF + RM/28.75)\, x_7 - (SF + RM/28.75)\, x_9$

(3) $\quad L = (2SF + RM/28.75)(y_7 - y_8) + (SF + RM/28.75)(y_{10} - y_9)$

(4) $\quad PM = 28.75\,(2SF + RM/28.75)\dfrac{x_7 - x_8}{z_8 - z_7} + 57.50\,(SF + RM/28.75)\dfrac{x_{10} - x_9}{z_{10} - z_9}$

Again, we may satisfy either (2) or (4) but not both, due to flexure stiffness combined with system yielding. The remaining interactions will be smaller with displacement control at the moment table, and with more flexible flexures.

Yaw System Adjustments. See Fig. 6

Since (14) is more accessible than (16), we will adjust (14) instead of (12).

First adjustment is to eliminate lift due to yaw. Assume that z_{16} is set correctly to make the two arms of the floating level equal (it does not matter if this assumption is false). Then if z_{14} is wrong, the loads in struts 13 - 14 and 15 - 16 will be unequal and their differences will give drag (and pitching moment). Let the yawing moment produce tension of magnitude F_1 in strut 13 - 14, compression of amount F_2 in strut 15 - 16, and a drag D.

Then taking moments about 15,

(1) $\quad YM = 13\, F_1 + 6.5\, D$

The sum of the forces must be zero, so that

(2) $\quad F_1 + D = F_2$

Finally, taking moments about the fulcrum of the reversing lever, and letting z_{14} be the error in position of 14,

(3) $\quad F_1\,(10 + z_{14}) = 10\, F_2$

These three equations determine F_1, F_2 and z_{14} when a known YM is applied and D is measured. The solution is

(4) $\quad z_{14} = \dfrac{130\, D}{YM - 6.5\, D}$

or since $D \ll YM$, approximately

(5) $z_{14} = 130 \dfrac{D}{YM}$

With drag caused by yawing moment eliminated, we can assume $F_1 = F_2 = YM/13$; then

(6) $PM = \dfrac{YM}{13} (y_{13} - y_{15})$

(7) $L = \dfrac{YM}{13} (y_{14} - y_{13}) + (y_{16} - y_{15})$

(8) $RM = 6.5 - \dfrac{YM}{13} (y_{13} - y_{14}) + (y_{16} - y_{15})$

(9) $SF = \dfrac{YM}{13} (z_{14} - z_{15}) + (z_{15} - z_{16})$

To satisfy (8) we may adjust either z_{13} or z_{15}; since to change z_{14} or z_{16} could reintroduce a drag due to yaw.

We will agree not to adjust y_{16}; then equations (6), (7) and (8) determine y_{13}, y_{14}, and y_{15}.

Eliminate $(y_{13} - y_{14})$ from (7) and (8), which gives

(10) $(y_{16} - y_{15}) = 6.5 L + R$

This enables $y_{16} - y_{15}$ to be computed. We will assume y_{16} is its nominal value from the drawings, then y_{15} can be computed.

From equation (6),

$$y_{13} = y_{15} + \dfrac{13 \; PM}{YM}$$

Finally from (7)

$$y_{14} = y_{13} - (y_{16} - y_{15}) + \dfrac{13 \; L}{YM}$$

In the case of the yawing moment system, all the interaction equations can be satisfied, and it should be possible to remove all traces of yawing moment in the other five components even with non-zero flexure stiffness and yielding of the system under yawing moments.

Force Applied	Forces Read					
	D	L	SF	PM	RM	YM
D	1	0	0	0	a_1	b_1
L	0	1	0	0	0	0
SF	0	0	1	c_1	0	d_1
PM	0	0	0	1	a_2	b_2
RM	0	0	0	c_2	1	d_2
YM	0	0	0	0	0	1

Due to flexure stiffness and yielding of the moment table under loads, not all the interactions can be made zero. We may make either $a_1 = a_2 = 0$ or $b_1 = b_2 = 0$, but not both. Similarly, we may make $c_1 = c_2 = 0$ or $d_1 = d_2 = 0$.

It is also possible to calibrate the system so that neither the a_i or b_i are zero, but the a_i are half the size they would be if the b_i were zero, and the b_i are half what they would be with the $a_i = 0$.

After the interactions have been eliminated, the calibration consists of adjusting two potentiometers for each of the six Brown Instruments (modified), one for the high range and one for the low range, so that the readings will represent pounds or inch pounds with perhaps a shift of decimal point. The adjustments to eliminate interactions assume that the Brown Instrument current-measuring potentiometers were roughly adjusted in the beginning.

Tare Moment Compensator Adjustment.

The tare moment compensator (See Fig. 8) is driven by steel tapes from the crescent, so that it rotates about its rotation center with the same angular velocity and direction as the crescent. To study its action, one can think of it as being translated so that its center of rotation coincides with that of the crescent, and then frozen to the crescent. The c.g. of the crescent, sting and model, together with the compensator moving mass thus translated and frozen to the crescent, will be called the combined c.g.

The weight W_1 moving parallel to the strut axis serves to move the combined c.g. in this direction, while W_2 serves to move the combined c. g. perpendicular to the strut axis. Both must be adjusted until the combined c.g. coincides with the axis of rotation.

Generally models have their c.g. on the strut axis; in this case changing models will only shift the combined c.g. along the axis, and hence require only readjustment of the weight W_1. For this reason, W_1 is motor-driver with remote control.

If w_2 is too high by a distance δ_2, then the combined c.g. is too high by a distance $r_2 = \delta_2\, w_2/W$ where W is the combined mass of compensator, model, crescent and strut. The resulting moment at angle α is $W\, r_2 \sin \alpha$ or

$$PM = w_2\delta_2 \sin \alpha$$

If w_1 is too far upstream by a distance δ_1, the resulting moment is $- W\, r_1 \cos \alpha$, where $r_1\, \beta\, \delta_1\, w_1/W$; or with moment zeroed when $\alpha = 0$,

$$PM = \delta_1\, w_1\, (1 - \cos \alpha)$$

All the bell-cranks transmitting forces to the pots have adjustable counterweights to enable static zeros to be set. In general, when a model is changed, only the counterweights on the lift lever and on the pitching moment bell-crank will need changing. These are remotely controlled by motors. But if the c.g. of the model is off the xy-plane, the roll moment counterweight will also need adjustment to zero this component.
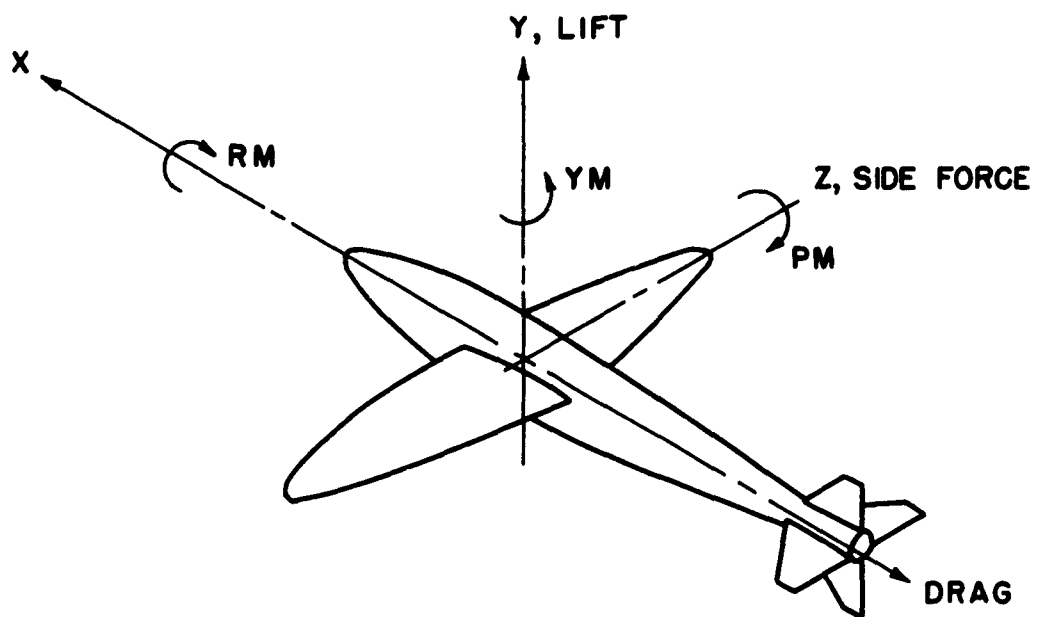
FIG. I

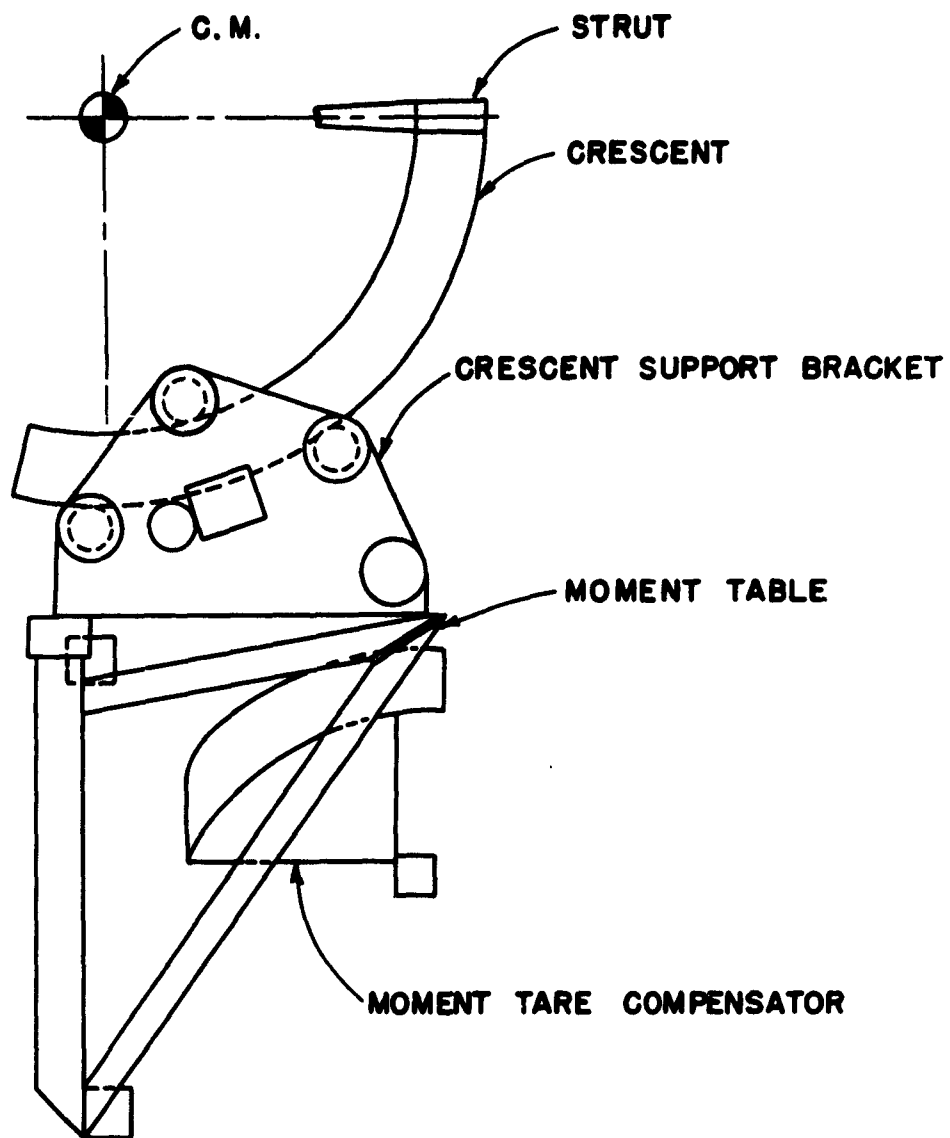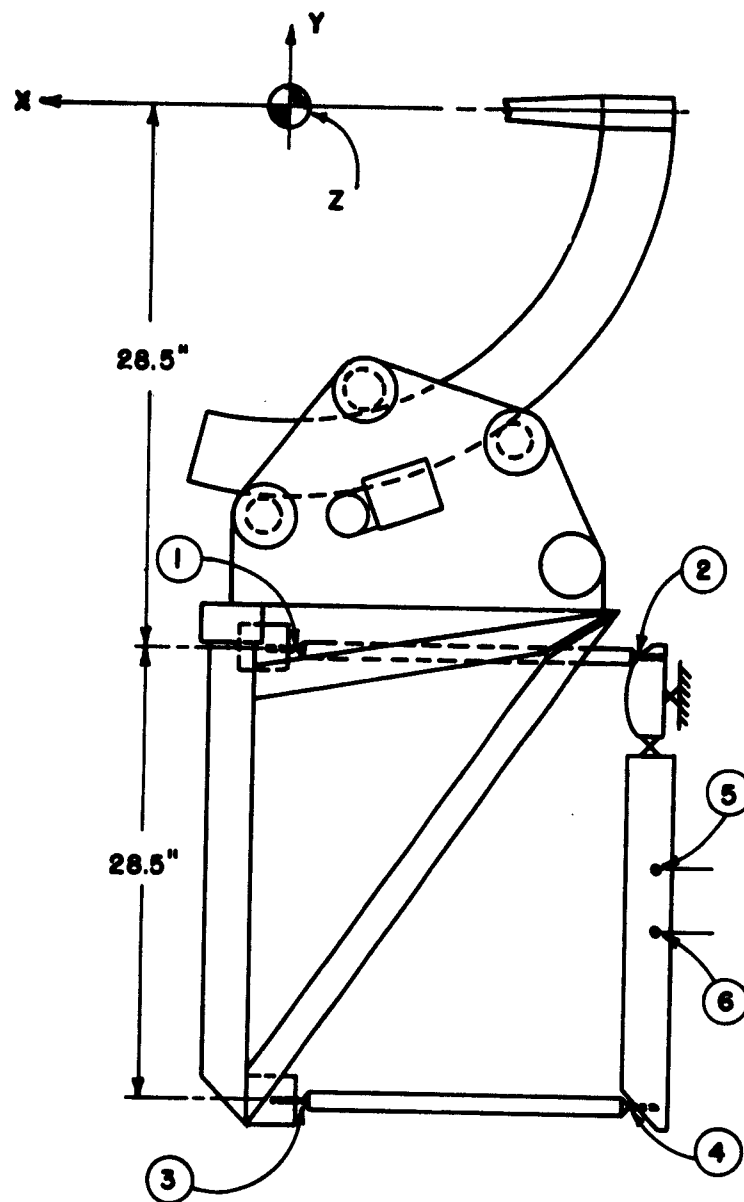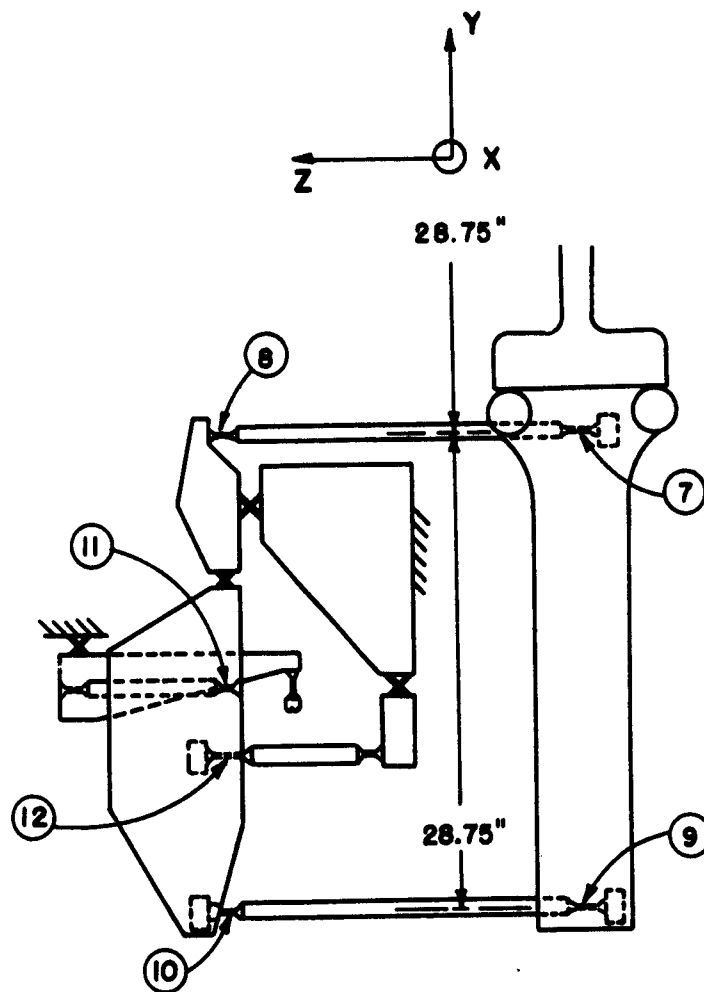SIGN CONVENTIONS FOR FORCES, DIRECTIONS & MOMENTS

FIG. 2

FLOATING PART OF BALANCE SYSTEM

ADJUSTMENTS: $Y_1, Z_1, Y_2, Z_2, Y_3, Z_3, Y_4, Z_4, Y_5, Y_6, \ell_{1-2}, \ell_{3-4}, \ell_5, \ell_6$
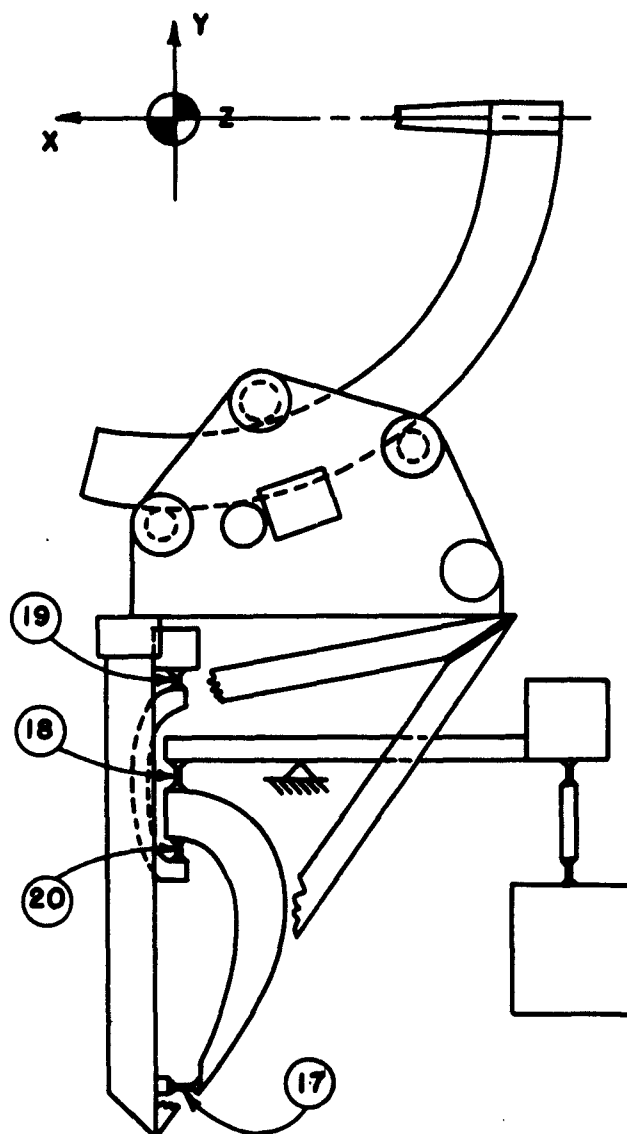
FIG. 3

DRAG & PITCH SYSTEM

ADJUSTMENTS

$$X_7, Y_7, X_8, Y_8, X_9, Y_9, X_{10}, Y_{10}, Y_{11}, Y_{12}, \ell_{7\text{-}8}, \ell_{9\text{-}10}, \ell_{11}, \ell_{12}$$
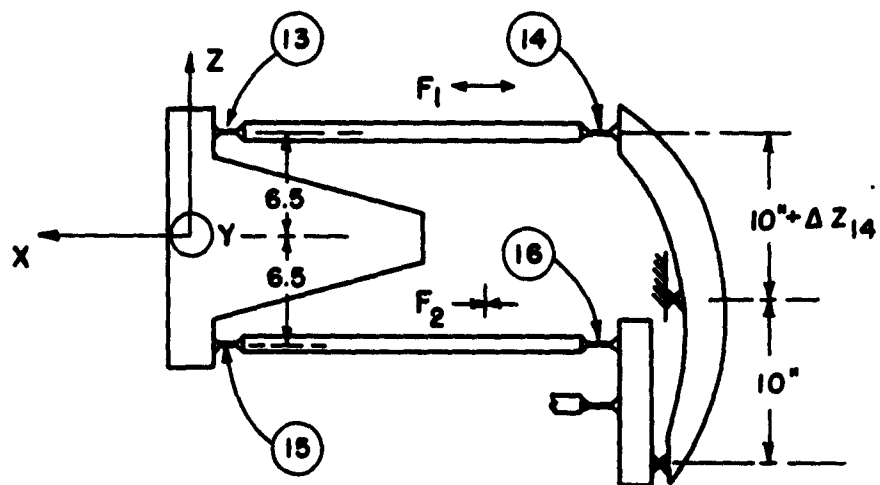
FIG. 4

SIDE FORCE & ROLLING MOMENT SYSTEM

ADJUSTMENTS NEEDED: 1 – LENGTH ADJUSTMENTS AT TOP OF ⑱
2 – X, Z   "   "   "   "   "
3 – X, Z   "   " ⑰ OR ⑲

FIG. 5

LIFT SYSTEM

ADJUSTMENTS: Z_{14}, Z_{13}, OR Z_{15}, Y_{13}, Y_{14}, Y_{15}

FIG. 6

YAW SYSTEM

ELIMINATION OF INTERACTIONS
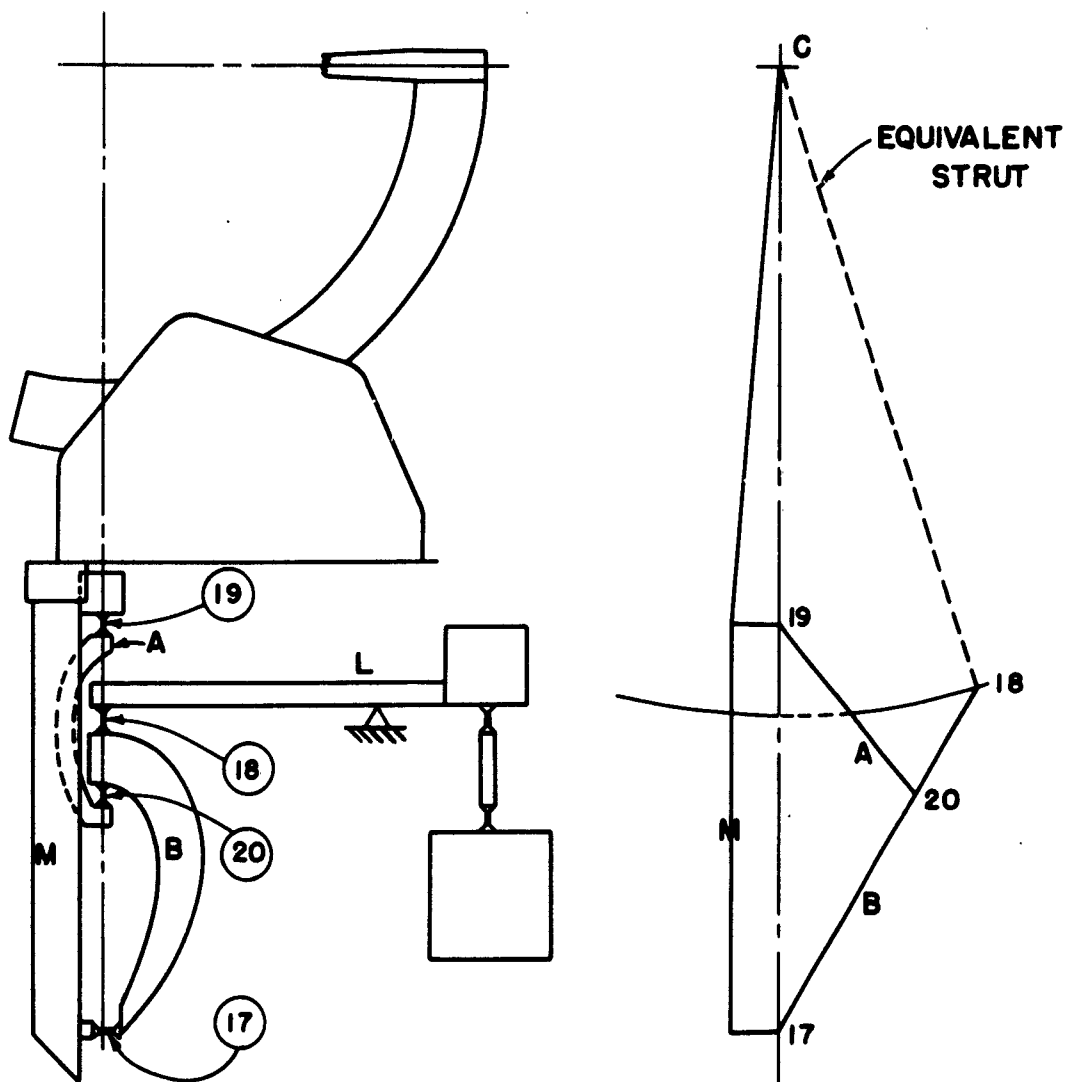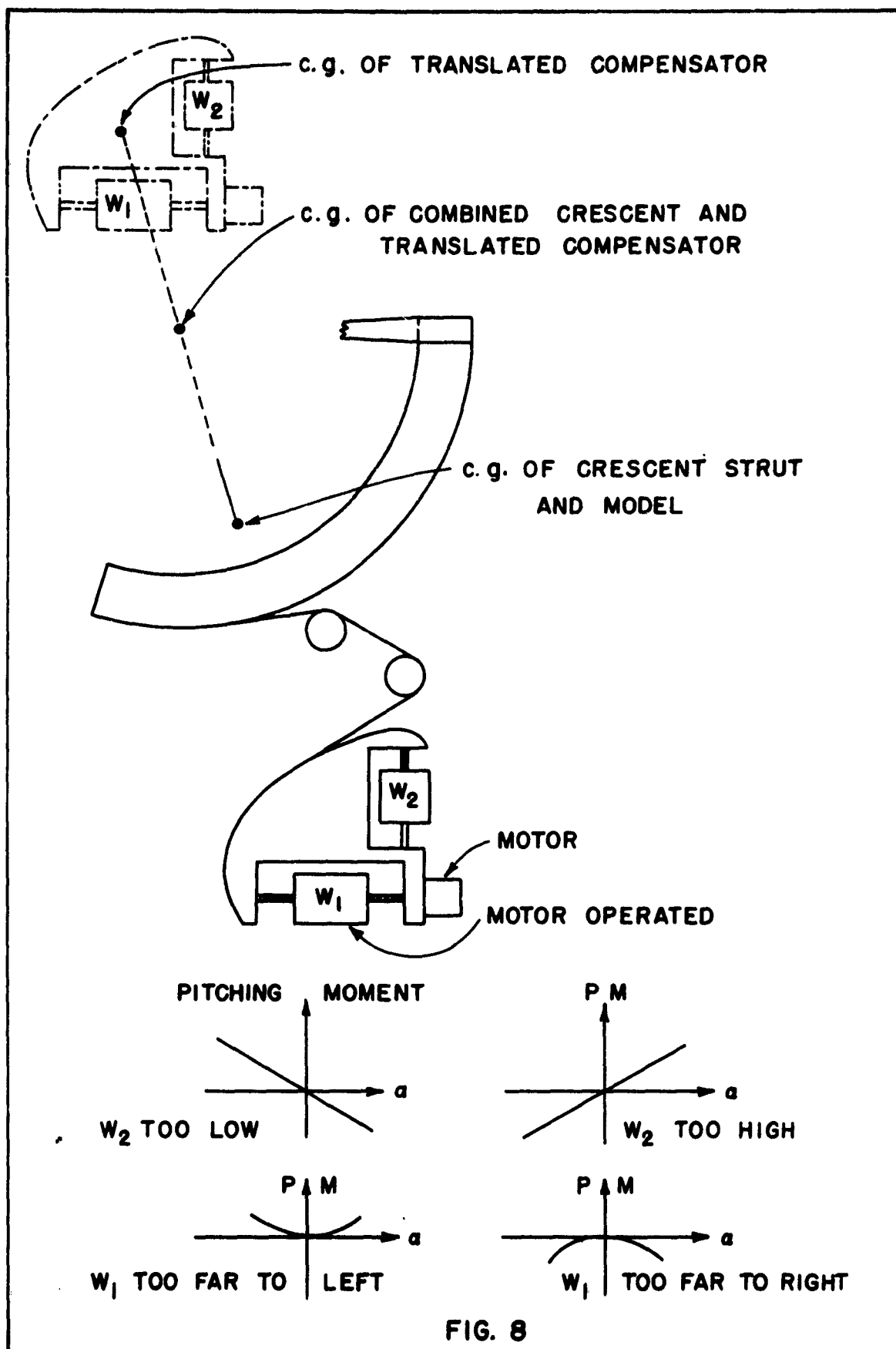
I. LIFT

DESCRIPTION OF LINKAGE

EQUIVALENT STRUT

FIG. 7

c.g. OF TRANSLATED COMPENSATOR

c.g. OF COMBINED CRESCENT AND
TRANSLATED COMPENSATOR

c.g. OF CRESCENT STRUT
AND MODEL

MOTOR

MOTOR OPERATED

PITCHING MOMENT

W_2 TOO LOW

P M

W_2 TOO HIGH

P M

W_1 TOO FAR TO LEFT

P M

W_1 TOO FAR TO RIGHT

FIG. 8

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